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DESIGN AND OPTIMIZATION OF CRANKSHAFT FOR FOUR CYLINDER 4-STROKE SPARK IGNITION ENGINE USING TRANSIENT STRUCTURAL ANALYSIS

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ABSTRACT

The objective of this article is to propose an optimized design structure of the crankshaft for a four-cylinder four stroke spark ignition engine. The new design is compared to the existing one on different grounds of performance characteristics for engines with varied purposes. A three-dimensional crankshaft was created through reverse engineering techniques, on SOLIDWORKS and then imported to ANSYS for a transient structural analysis. The variable time varying force was derived from the pressure-crank angle graph of the CBR600 engine and used ultimately for this analysis. Three new materials were tested on the crankshaft – 4130 chromoly steel, Titanium alloy Ti6Al-4V, Epoxy Carbon fibre (230 GPa- Woven). The focus is on the new light weight design of the crankshaft incorporating the usage of the abovementioned materials and through transient structural analysis, the feasibility is studied. The reduced weight of the crankshaft was 7.6 Kg and 12.9 percent less compared to the existing crankshaft without compromising the strength to weight ratio.

KEYWORDS: Crankshaft Optimization, Transient Structural Analysis, Weight Reduction & New Materials

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Original Article

1. INTRODUCTION

Crankshaft is a vital part of the internal combustion engine (Montazersadgh et al). It has the most important function of transforming the energy produced by the internal combustion to the rotatory energy of the wheels. It converts the reciprocating motion of the piston to a rotatory motion with a slider crank mechanism. It undergoes a large number of load cycles and variation in the forces by the connecting rods, hence the durability and fatigue life have to be taken as an important design consideration. One end of the crankshaft is attached to the flywheel which is responsible for transmitting the power to the clutch and ultimately the gear train. Hence to achieve a good fatigue strength with better fuel efficiency and higher output, an optimised design is proposed.

There have been multiple studies on this subject and all have resulted in viable conclusions.

This article (M.JIAN et al) showed the maximum stress, deformation and high-risk areas though a static structural and model analysis on four-cylinder engine crankshafts.

This methodical paper (THIRUNAVUKKURASU et al) proposed various ways to reduce the weight of the crankshaft. He suggested the use of hollowed out crank pins and journals which witnessed an increase in the safety of factor and almost a quarter reduction in the weight.

Whereas(P. Ciitti et al), the importance of crankshafts is discussed with a focus on material selection and manufacturing. The objective of the paper is studying the usage of newer materials and production technologies for

this vital component of the engine. In this article, materials 38MnVS6 and 48MnVS3 are studied with surface hardening techniques and thermal treatments.

In this article(Ramnath et al), a method is devised to derive various parameters of the development of a crankshaft. CATIA and reverse engineering techniques were used to design a computer aided model and then optimization analysis is used to investigate the fatigue performance of ductile cast iron, forged steel and aluminium alloy. A Finite element analysis is carried out and a probable weight reduction is suggested for the forged steel material which is feasible after weight reduction.

In the many articles in this discussion, various methods are suggested to reduce the weight of a crankshaft. This paper works toward a weight reduction in crankshaft design through reverse engineering, design optimization and variation of materials, frequency analysis used in order to increase the strength to weight ratio and the engine performance. This particular amalgamation of techniques proposes a novel method in this paper to compare the crankshaft design on various parameters.

1.1 Engine Specification

The engine used in this article is CBR600-f4i, the specification of the specific engine used is provided in the following table 1

Table 1: Engine Specification

Bore (mm)	67
Displacement in CI	36.6
Starter	Electric
Valve Configuration	Double Overhead
Compression Ratio	12:1
Displacement in cc	599
Engine Type	Horizontal In-line
Stroke Length	42.5

Figure 1: CBR600f4i Engine Crankshaft.

1.2 Materials

High strength, ductility and fatigue resistance are the critical properties required for the material selection of a crankshaft. Chromoly steel is actually an alloy of steel grade 4130[6]. As evident by the grade number, it has approximately .3% carbon content by weight. The additional alloying elements found in AISI 4130 help in increasing the strength, which can be further enhanced using hardening process. The added Chromium helps increasing the corrosion resistance and

hardenability. The added molybdenum results in increased toughness. It also can be easily work hardened, carburizing the alloy can also give case hardened chromoly.

Titanium alloys are widely used due to their high strength, low weight ratio and great corrosion resistance [7]. It has led to a wide and diversified range of successful applications which demand high levels of reliable performance in various fields. Ti6Al-4V is the most widely used Titanium alloy. It features good machinability and outstanding mechanical properties. It is widely used in weight reduction applications in aerospace, automotive and marine equipment.[8].

Reinforced Carbon fibre epoxy composites are used widely in the automotive sector due to High strength to weight ratio, corrosion resistance, fatigue resistance and light weight. It is also chemically stable and hence does not react with different surrounding elements. The use of carbon fibre in crankshaft is studied as a possibility in this article comparing it with metal alloys and checking its feasibility.

2. METHODS

2.1 Solid Modelling

The initial model of the crankshaft was developed using a state-of-the art 3D Scanner Steinbichler Comet L3D Blue Light Scanner with a resolution of 2Mpx spanning over 1600x1200 pixels giving accurate dimensions and parameters. The crankshaft used was of a CBR600F4i engine. The initial scanned model and CAD model before the parametric optimization are as follows

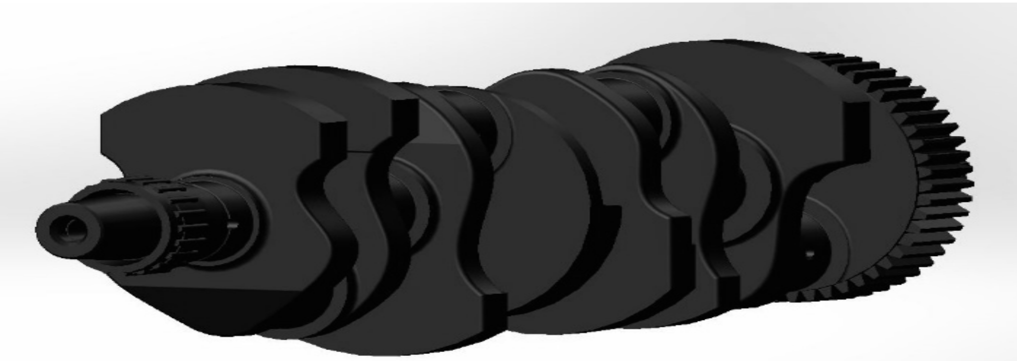

Figure 2: 3D Scanned Model.

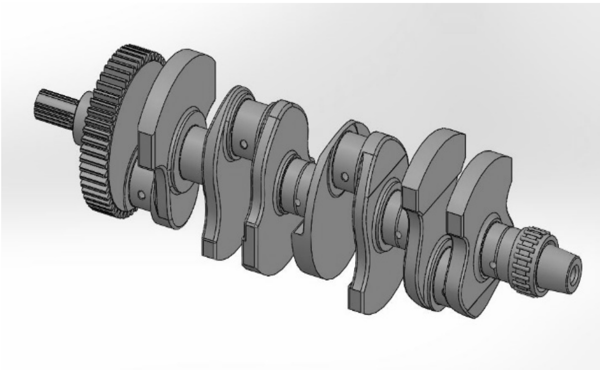

Figure 3: CAD Mode.

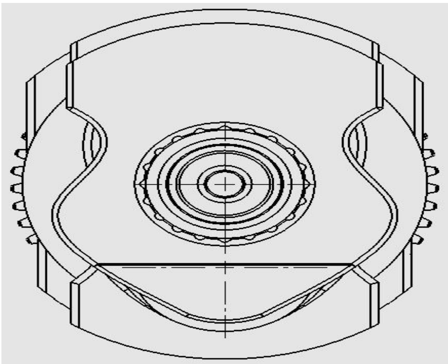

Figure 4: End View Drawing.

2.2 Finite Element Analysis

After the scanning process, development of the Computer Aided Design Model in SolidWorks 2017, the model is then imported to ANSYS Workbench v18.2 for transient structural analysis. The pressure forces subjected to different parts of the crankshaft are derived from the Pressure- Crank angle diagram as shown in figure 5.

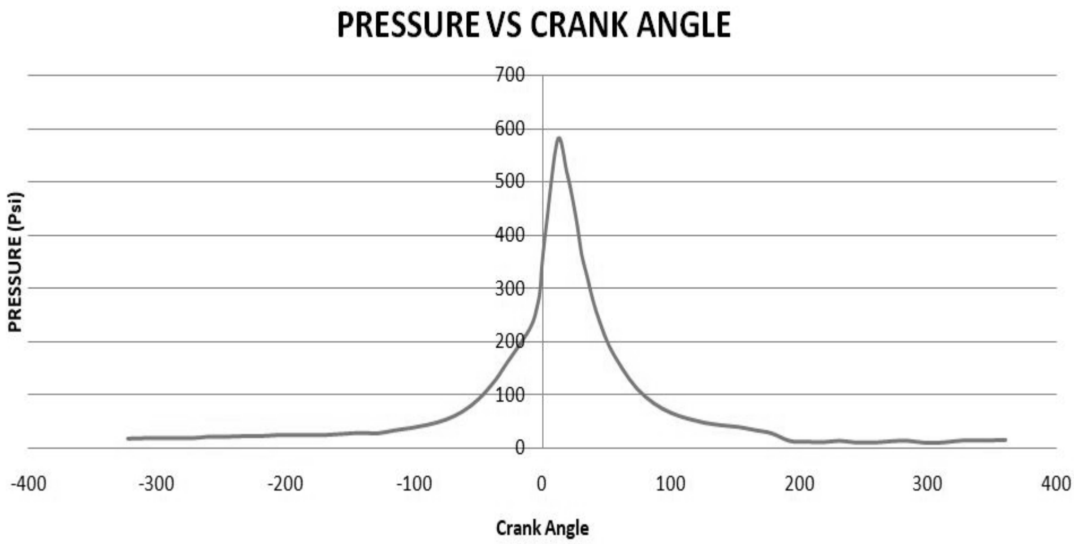

Figure 5: Pressure-Crank Angle Graph.

The total force subjected to the piston is the combination of pressure and inertial forces, which are both functions of the crank angle. Friction forces have been neglected in this study.

$$F(\theta) = F_p(\theta) - F_i(\theta) \tag{1}$$

Where θ= Crank Angle

F_p= Pressure Force

F_i= Inertial Force

Hence, the moment derived from these forces as shown in Equation (1) is also a function of the crank angle which were calculated and plotted with the help of simple trigonometric relations and MATLAB. The Turning moment (N-m) vs the crank angle (degrees) is shown in Figure 6. The four graphs of every cylinder each are superimposed to give better informative data. The consecutive peaks correspond to Cylinder 1–2–4–3 respectively according to the firing order.

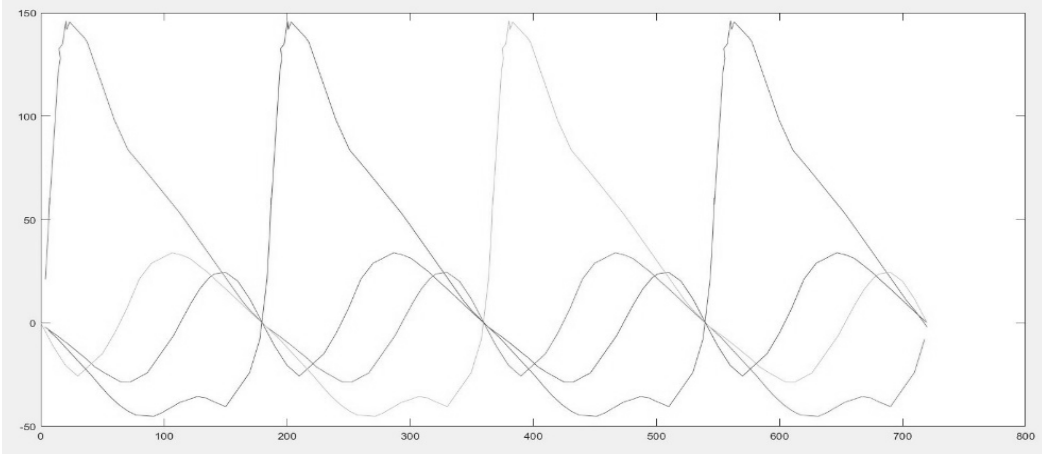

Figure 6: Turning Moment Diagram.

As we can see the maximum pressure is 580psi, which is transmitted to the crankshaft through the connecting pins. The model in the workbench environment is shown in Figure 7. The number of elements is 140,653 as shown in Figure 7. The following figures shows the analysis results of the original unmodified crankshaft under consideration, the moment were applied at the eccentricities as a function of crank angle obtained through MATLAB, cylindrical support option at bearing region and the rotational velocity applied at the output shaft was 314 rad/s. Figure 8 shows the equivalent stress at the crankshaft resulting in the maximum value of 100.6 MPa. Figure 9 shows the maximum deformation of 4.81e-6 metres at the crankpin region. Figure 10 shows the maximum shear stress according to the applied parameters, which comes out to be 6.27e6 Pa at the bottom fillet region of the crankshaft. Figure 11 shows the range of factor of safety of the analysis which was minimum 2.48 and maximum 15.

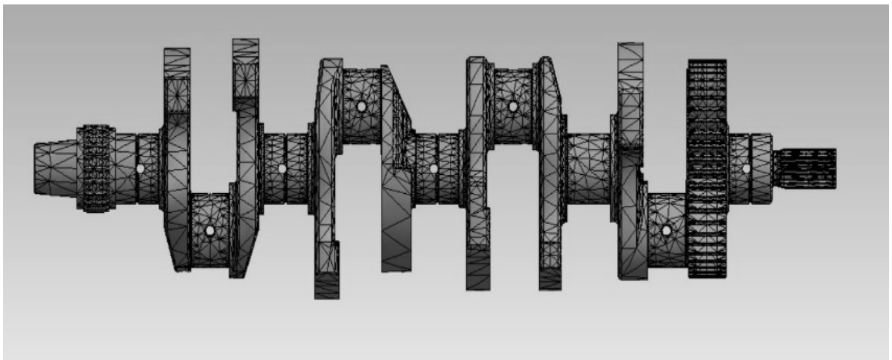

Figure 7: Crankshaft Mesh Model.

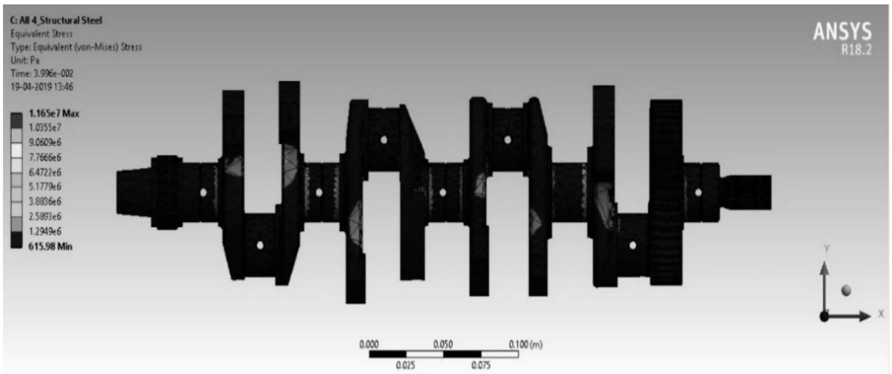

Figure 8: Equivalent Stress.

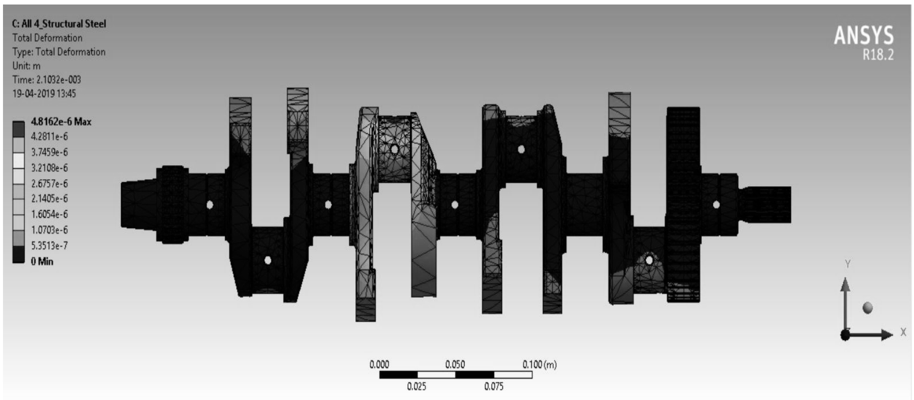

Figure 9: Total Deformation.

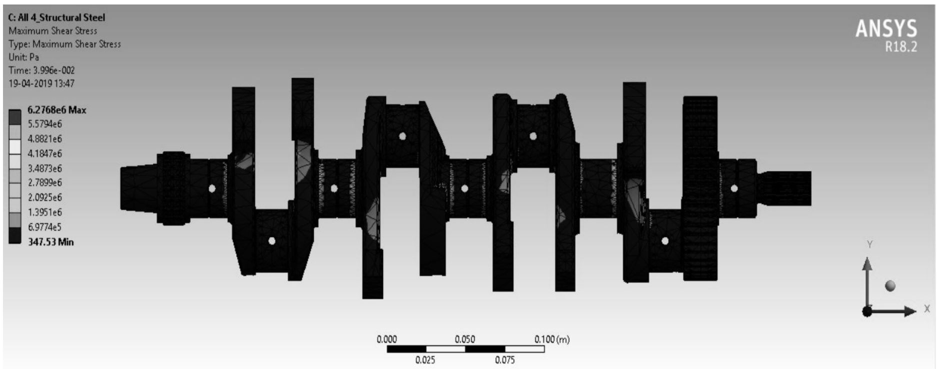

Figure 10: Maximum Shear Stress.

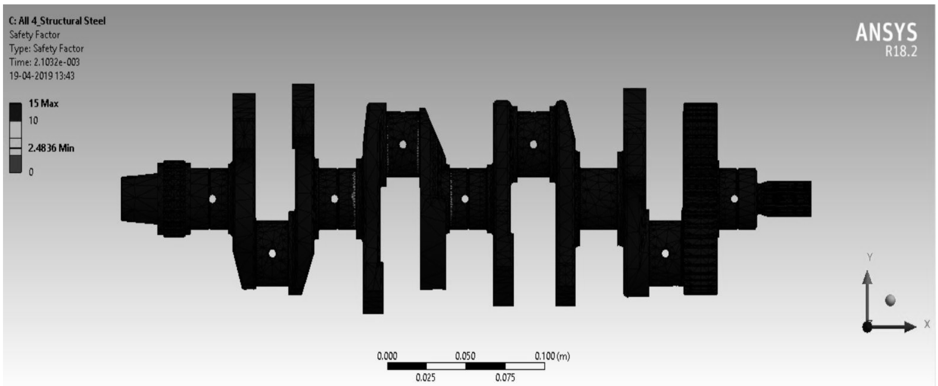

Figure 11: Factor of Safety.

2.3 Parametric Optimization

By careful investigation of the stress profiles and contours of the ANSYS results of the original Initial model of the crankshaft[9]. It is evident that some components of the crankshaft experience relatively lower stress values. A crankshaft is a vital mass-produced component of the internal combustion engine and any design optimization that could lead to a reduction in weight would result in significant advantages. The advantages range from increased fuel efficiency and reduced production cost. The only constraints being it does not demand a design change in the mounting locations and points of the engine block or the connecting rod.

Various research articles have suggested possible stress and weight reductions to further optimise the existing crankshaft design. After a careful study of the stress contours, it is evident that the counterweights, crank rod bearings and crank throw are experiencing lesser stress. The counterweights are for dynamic balancing and hence cannot be modified, even if they are experiencing very less stress values. The only solution is axi-symmetrically removing mass which will result in a static and dynamic balanced optimised design.

As it is shown in figure 12 and figure 13, there is a hole of 14.5 mm diameter in the crankpin region of the crankshaft. The hole goes throughout the length of the crankshaft and does not affect the functioning, and helps considerably in achieving appreciable proper weight reduction. To reduce the stress concentration at the edges, the fillet radius has to be increased. An increase in the fillet radius at the knuckle area of connecting rod and crank arm journal does not affect the geometry.

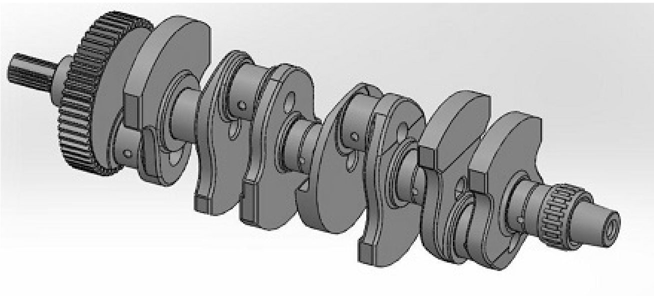

Figure 12: Design Optimization.

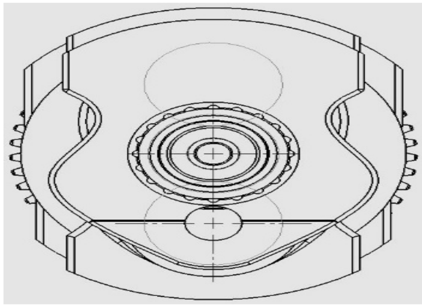

Figure 13: Side View.

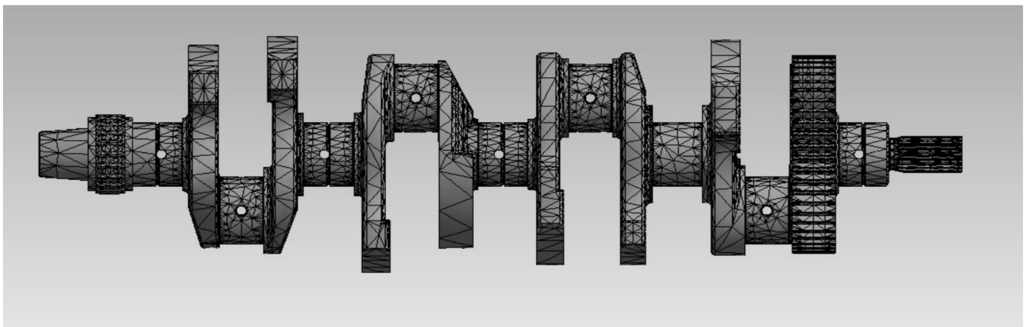

Figure 14: Crankshaft Mesh Model.

The following figures show the analysis results of the modified crankshaft under consideration. The moment was applied at the eccentricities as a function of crank angle obtained through MATLAB, cylindrical support option at bearing region and the rotational velocity applied at the output shaft was 314 rad/s. Figure 15 shows the equivalent stress at the crankshaft resulting in the maximum value of 90.59 MPa. Figure 16 shows the maximum deformation of 5.01e-6 metres at the crankpin region. Figure 17 shows the maximum shear stress according to the applied parameters, which comes out to be 5.22e7 Pa at the bottom fillet region of the crankshaft. Figure 18 shows the range of factor of safety of the analysis which was minimum 2.75 and maximum 15.

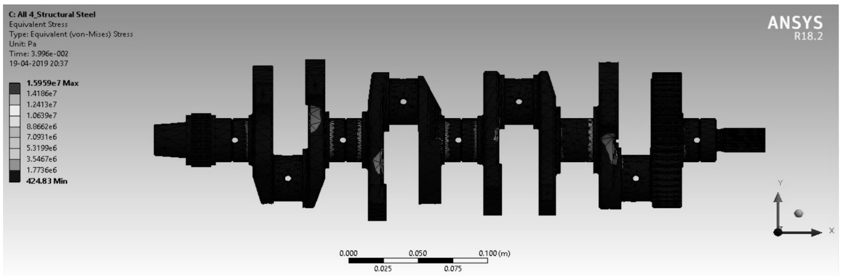

Figure 15: Equivalent Von-Mises Stress.

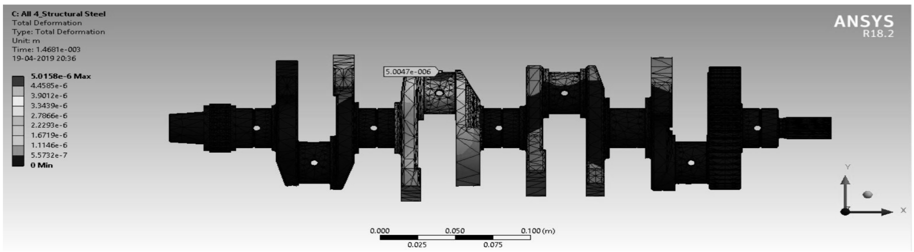

Figure 16: Total Deformation.

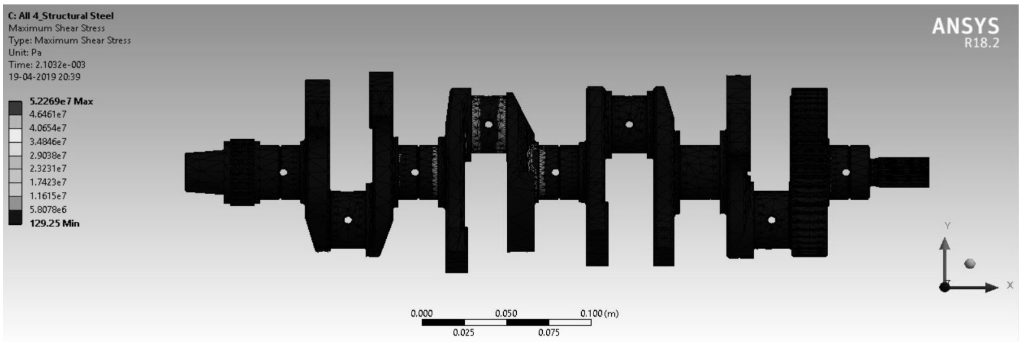

Figure 17: Maximum Shear Stress.

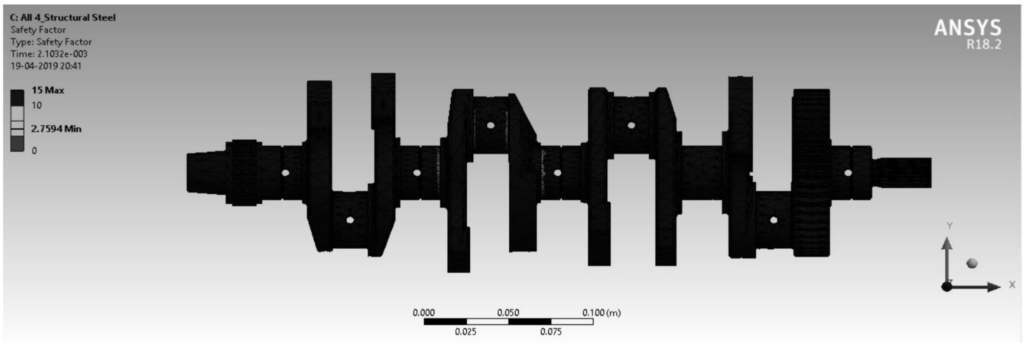

Figure 18: Factor of Safety.

3. RESULTS AND DISCUSSIONS

The following parameters including the maximum von-mises stress, total deformation, maximum shear stress and the factor of safety were found out using the finite element analysis. After parametric optimization, the maximum localized stress area occurs near the bottom fillet as observed in the graphical results obtained. Figure 17 shows the comparative equivalent stresses for all the four materials on the same scale.

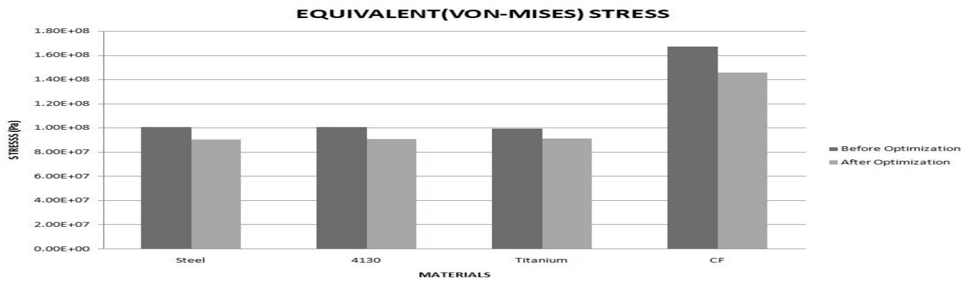

Figure 19: Comparison of Equivalent (von-mises) Stress between Original and Optimized Model.

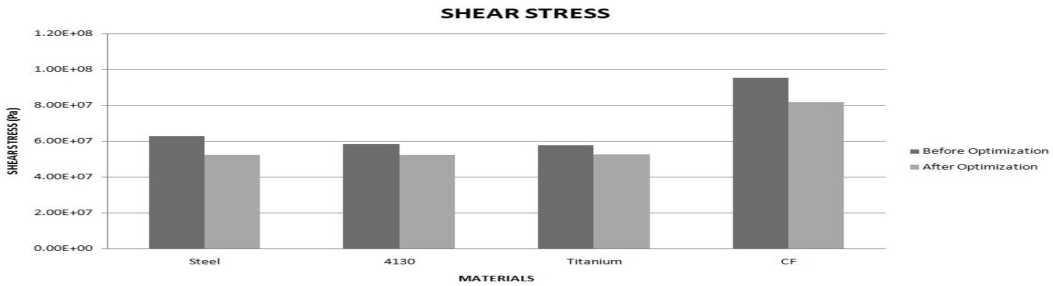

Figure 20: Comparison of Shear Stress between Original and Optimized Model.

With this detailed comparative study of different material applications, it was noted from Figure 19 that the most reduction in the equivalent stress value was observed in the Carbon Fibre Epoxy valued at 12.78%. Steel showed a reduction of 10% followed by 4130 Chromoly at 9.9%. The least reduction was showed by Titanium at 8.56%. Similarly, in Figure 20, we observe that the shear stress values are also reduced as a result of the parametric optimization. Steel showed the most reduction valued at 16.7% followed by Carbon Fibre epoxy, 4130 Chromoly and Titanium. Total deformation also reduces after the optimization.

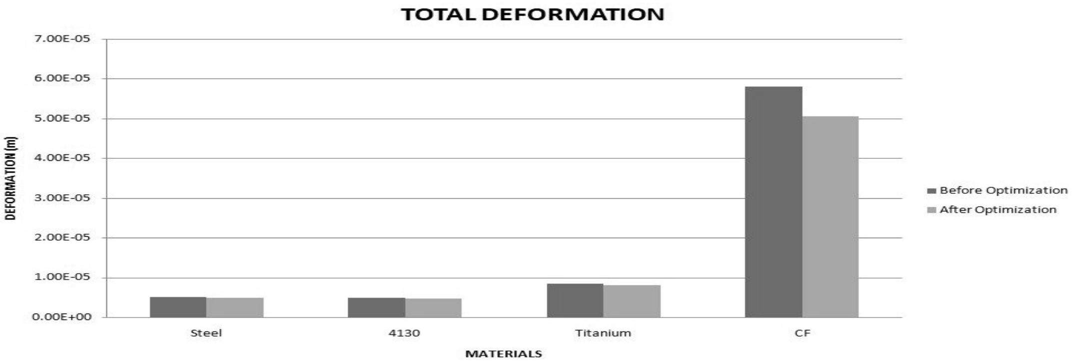

Figure 21: Comparison of Total Deformation between Original and Optimized Model.

As the engine under consideration is a 4-cylinder 4-stroke engine, the stress and strain values differ with position and time depending on the individual crank angles of the cylinders.

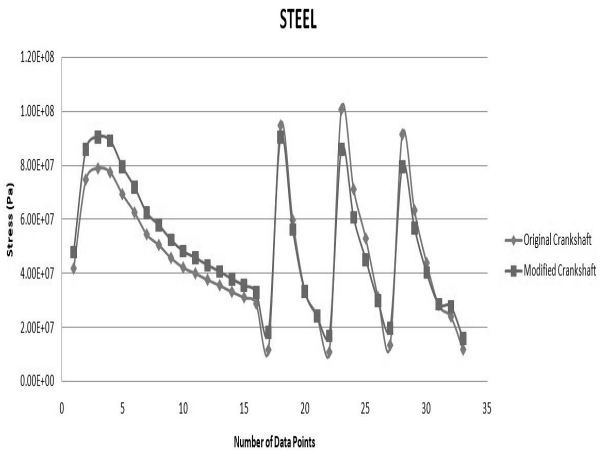

Figure 22: Comparison of Stress for Steel.

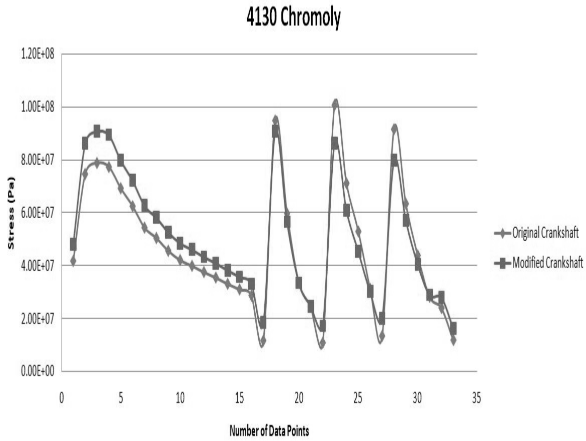

Figure 23: Comparison of Stress for 4130 Chromoly.

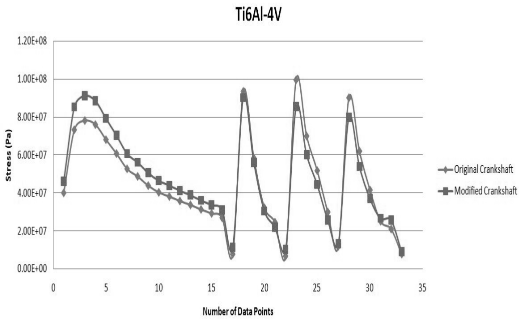

Figure 24: Comparison of Stress for Titanium.

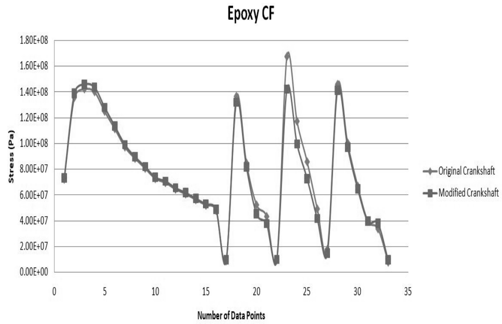

Figure 25: Comparison of Stress for Epoxy CF.

As we observe in the above stress vs number of data points graphs obtained, the moderate to higher time step values have decreased values of stress. So, the results are favourable in the high-performance ranges of the engine, thus it has wider acceptability and application in high RPM engines.

4. CONCLUSIONS

Undergoing this transient structural analysis, different results were obtained for the crankshaft and are compared in terms of equivalent (von-mises) stress, total deformation, shear stress and the factor of safety. By careful analyses of these results, possible locations were identified for mass removal to optimize the existing geometry of the crankshaft. Though there are problems that are identified and need solutions. This optimized design cannot be feasibly manufactured by conventional manufacturing techniques like casting, moulding etc. Hence additive manufacturing methods are advised to give better outputs. The second problem pertains to the high localized stress value close to the bottom fillet region of the crankpin, this problem can be solved by either increasing the surface area or by providing suitable fillets.

The weight was reduced by performing the parametric optimization of the crankshaft which is subjected to varying tensile and compressive loads. The crankpin region offered the greatest possibility for weight reduction and this opportunity is assessed in this article. A further analysis can be extended to a modal analysis to calculate the resonant frequency of the crankshaft and undertake a comparative study.

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